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PERFORMANCE OF R-22, R-407C AND R-410A AT CONSTANT COOLING CAPACITY IN A 10.0 SEER 10.5 kW RESIDENTIAL CENTRAL HEAT PUMP

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ABSTRACT

The performance of two long-term replacements R-407C (HFC-32/125/134a (23%/25%/52%)) and R-410A (HFC-32/125 (50%/50%)) was compared to R-22 on a constant compressor capacity basis in a 10.5 kW residential central heat pump. The performance evaluations were carried out in a calorimetric test facility using the Canadian Standards Association (CSA) / Air-Conditioning and Refrigeration Institute (ARI) rating conditions. The hermetic reciprocating compressor that was supplied with the heat pump was replaced with an open drive-reciprocating compressor connected to a variable speed motor. The performance of R-22, R-407C, and R-410A was measured using the same open drive-reciprocating compressor. The compressor speed was adjusted for each of the two HFC refrigerants to provide the same cooling capacity as the R-22 base case at the 'A' test condition.

Additional air-conditioning performance testing was completed on all the refrigerants at an extreme outdoor ambient temperature of 45°C to determine if the cooling capacity and EER of R-410A deteriorate at these temperatures compared to R-407C.

Performance characteristics were measured including system operating conditions, compressor shaft power, cooling and heating capacity, cooling energy efficiency ratio (EER) and heating coefficient of performance (COP).

INTRODUCTION

Previous studies of R-22 and long-term replacements R-407C and R-410A have used different types and sizes of compressor to directly compare the capacity and performance differences of these refrigerants (Linton 1996). The objective of this investigation is the comparison of these three refrigerants using the same compressor operating at different speeds (supplying the same cooling capacity) to provide a more accurate indication of the system differences of the long term replacements. The comparison of R-407C and R-410A was completed in a 10.5 kW Carrier residential central heat pump with the base case R-22 at the standard Air-Conditioning and Refrigeration Institute (ARI) / Canadian Standards Association (CSA) rating points (CSA 1991). The air-conditioning performance at a high outdoor ambient temperature (approximately 45°C DB) was also evaluated to determine if the cooling capacity and EER of R-410A deteriorate at these temperatures compared to R-407C.

TEST DESCRIPTION

The performance evaluation was completed in the Calorimetric Test Facility located at the Thermal Technology Centre, National Research Council. The Calorimeter consists of two environmentally controlled test chambers that simulate indoor and outdoor conditions with precise control of the air dry bulb and wet bulb temperatures.

In order to compare the performance of R-407C and R-410A with R-22, a series of tests was performed on a standard 10.0 SEER Carrier 10.5 kW cooling capacity air-to-air residential heat pump. The original hermetic reciprocating compressor that came with the heat pump was changed to an external two cylinder-reciprocating compressor driven by a 10 HP variable speed electric motor. Original equipment on the heat pump also included fixed orifice type expansion devices on the indoor and outdoor coil. The indoor coil consisted of a single sloped fin and tube coil with three tube rows and five refrigerant circuits. The indoor coil was mounted in the original position as received from the factory. The indoor coil was in a cross-parallel flow configuration for the air-conditioning tests and in a cross-counter flow configuration for the heating tests. The outdoor coil was a single row fin and tube type coil with three refrigerant circuits. Table 1 lists the five standard CSA indoor and outdoor test conditions (plus the extreme operating condition) for a split system air-to-air heat pump.

Type of Test	Indoor Conditions	Outdoor Conditions
"A" Steady State Wet-Coil	27°C DB / 19°C WB	35°C DB
"B" Steady State Wet-Coil	27°C DB / 19°C WB	27.8°C DB
Maximum Oper. Cond. (AC)	27°C DB / 19°C WB	40°C DB
Extreme Oper. Cond. (AC)	27°C DB / 19°C WB	45°C DB
High Temperature Heating	21°C DB / 16°C WB	8.3°C DB / 6.1°C WB
Low Temperature Heating	21°C DB / 16°C WB	-8.3°C DB / -9.4°C WB

Table 1: CSA / ARI cooling test conditions for split system residential air-to-air heat pumps

The heat pump was extensively instrumented, and the air enthalpy and refrigerant mass flow rate methods were used to determine the unit's indoor coil steady state cooling and heating capacity. Energy balances between the air and refrigerant side were within 1% to 2% for R-22 and within 2% to 3% for R-407C and R-410A. It is believed that the larger discrepancies in the energy balance for R-407C and R-410A are related to uncertainties of the refrigerant properties. The cooling and heating capacities reported in this paper were from the air side measurement data. Refrigerant temperatures were recorded using type T (copper-constantan) thermocouples soldered to the refrigerant tubing. The uncertainty of the thermocouple temperature measurements was ±0.6°C. Refrigerant pressures were measured using pressure transducers connected to static pressure taps located at strategic points in the system. The pressure transducers were calibrated to ±3 kPa. Refrigerant mass flow was measured directly with a Coriolis effect mass flowmeter mounted in the liquid line leaving the condenser. The mass flow meter was calibrated to provide an accuracy of ±0.5% of measurement. Shaft power input to the reciprocating compressor was measured with a torque and speed sensor that was mounted between the drive shaft and the electric motor. The torque sensor is a strain gauge type with an accuracy of $\pm 5.6 \times 10^{-2}$ Nm. The indoor and outdoor fan power was measured with watt/VAR transducers with an accuracy of ±1.0% of reading, and the supply voltage to the compressor and indoor and outdoor fans was regulated at 230 volts.

The energy efficiency ratio (EER) and coefficient of performance (COP) figures reported in the paper were based on the shaft power consumption of the compressor and the electrical power consumption of the indoor and outdoor fans. The indoor air quantity was set so that a minimum external resistance of 37.5 Pa was maintained at the outlet of the unit by adjusting an auxiliary fan located on the outlet of the test section ductwork.

For the R-22, R-410A and R-407C performance tests the only changes made to the heat pump were to substitute an external variable speed reciprocating compressor for the hermetic compressor and the installation of electronic expansion valves (EEV) and bypass check valves in place of the factory supplied combination fixed orifice and check valves. The EEV allowed accurate setting of the superheat for each refrigerant.

To achieve a fair comparison of a zeotrope to a single refrigerant or near azeotrope, the refrigerant cycle operating conditions need to be defined. The evaporating temperature was defined as the mean of the evaporator outlet pressure dew point and the evaporator inlet temperature. The condensing temperature was defined as the mean of the dew point and bubble point at the average condensing pressure. The superheat was measured from the compressor inlet pressure dew point and the subcooling temperature from the expansion valve inlet pressure bubble point respectively. The source of thermodynamic properties for R-22, R-407C, and R-410A was REFPROP version 4.0.

TEST METHOD

The compressor speed, refrigerant charge and EEV orifice setting was adjusted for each refrigerant at the "A" air-conditioning test conditions to achieve a 10.5 kW capacity and maximum EER. For the other three air-conditioning operating test conditions the compressor speed and refrigerant charge were not adjusted. The expansion valve was set in a manual operating mode at the same orifice setting that was used for the "A" test condition to simulate the air-conditioner operating with a fixed orifice expansion device. The same compressor speed and refrigerant charge were used for the heat pump in the heating mode. The EEV orifice setting was set to obtain the maximum cooling capacity at the high temperature heating test condition. The same orifice setting was used for the low temperature heating test condition. Setting the EEVs in this way closely duplicates the operation of the heat pump with the fixed orifices that the heat pump was originally equipped with. Using the established refrigerant operating charge and fixed orifice setting for the evaporator superheat at the "A" test condition, the performance of the heat pump was evaluated for the specified indoor and outdoor test conditions.

The baseline performance tests were completed for R-22 using a polyol ester (POE) lubricant with a viscosity of 32 mm^2/s at 40°C. The performance tests were then repeated with R-407C and R-410A and the same POE lubricant used in the R-22 tests. Finally, the compressor was operated in a vacuum at the same compressor speed that was used for each refrigerant and the same POE lubricant to measure the mechanical losses of the compressor.

TEST RESULTS AND DISCUSSION

Effect of Compressor Speed on Performance

The heat pump constant cooling capacity tests was run using the same variable speed-reciprocating compressor for all the test refrigerants. Table 2 shows the compressor RPM required to maintain the same cooling capacity for each refrigerant at the "A" test condition. The Table shows that R-410A required a 43% lower compressor speed than R-22 for the same cooling capacity. The Table also shows the required shaft power (mechanical losses) of the compressor when it was operated in a vacuum at the test operating speeds that were used for each refrigerant. By factoring in these mechanical losses it is estimated that the energy consumption of R-410A would be about 3% higher at the "A" test condition if the compressor speed was similar to the two other refrigerants.

	R-22	R-410A	R-407C
Compressor speed for tests (RPM)	1683	948	1725
Compressor power in a vacuum at test speed (watts)	147	65	155

Table 2: Compressor operating speed and power in a vacuum

The lower compressor speed used for R-410A will also have an effect on the compressor volumetric and isentropic efficiency. Work on reciprocating compressors by Villadsen (1985) showed that volumetric efficiency would be higher at lower compressor speeds. More importantly for compressor power the lower compressor speed could provide a significant improvement in the isentropic efficiency. The lower compressor operating speed required by R-410A therefore provided it with an unfair efficiency advantage over R-22 and R-407C. However, no correction factor was applied to the performance test results shown for R-410A.

Comparison of Operating Conditions

Evaporator Outlet Pressure

Cooling mode During testing at the cooling mode operating conditions R-22 and R-407C had similar evaporator outlet pressures, with R-407C having a slightly lower evaporator pressure. R-410A is a higher-pressure refrigerant than R-22 and it had evaporator outlet pressure that was about 64% higher that R-22.

Heating mode R-407C had a slightly higher evaporator outlet pressure than R-22 for the two heating mode test conditions. R-410A had evaporator outlet pressures that ranged from 71% - 76% higher than R-22.

Condenser Inlet Pressure

Cooling mode The condenser inlet pressure of R-407C was 12% to 15% higher than R-22 for the four cooling test conditions. R-410A had a condenser inlet pressure that ranged from 54% to 58% higher than R-22.

Heating mode The condenser inlet pressure of R-407C was 5.4% to 21% higher than R-22 for the two heating mode test conditions. R-410A had a condenser inlet pressure that ranged from 56% to 69% higher than R-22

Compressor Pressure Ratio

Cooling mode The compressor pressure ratio of R-407C ranged from 12% to 14% higher than R-22 for the air-conditioning conditions. R-410A had the lowest compressor pressure ratio of the refrigerants tested, ranging from 9% to 12% lower than R-22. The measured evaporating and condensing temperatures of all the refrigerants were similar.

Heating mode The compressor pressure ratio of R-407C was 0.7% to 9.3% higher than R-22 for the two heating mode test conditions. R-410A had the lowest compressor pressure ratio of the refrigerants tested ranging from 3.4% to 11.7% lower than R-22. The measured evaporating and condensing temperatures of all the refrigerants were also similar in the heat pump heating mode, with R-407C having the highest evaporating temperature and R-22 the lowest.

Compressor Discharge Temperature

Cooling mode The compressor discharge temperature was measured at the outlet of the compressor. For air-conditioning test conditions R-407C had a 3°C to 5°C lower discharge temperature than R-22. R-410A had compressor discharge temperatures that were 9°C to 11°C lower than R-22.

Heating mode R-407C had compressor discharge temperatures that were 1°C to 12°C lower than R-22 for the heat pump test conditions, R-410A's discharge temperature was about 1°C to 5°C lower than R-22. The lower discharge temperatures of the R-22 replacements could provide an advantage to compressor and lubricant long-term operation when at extreme operating conditions.

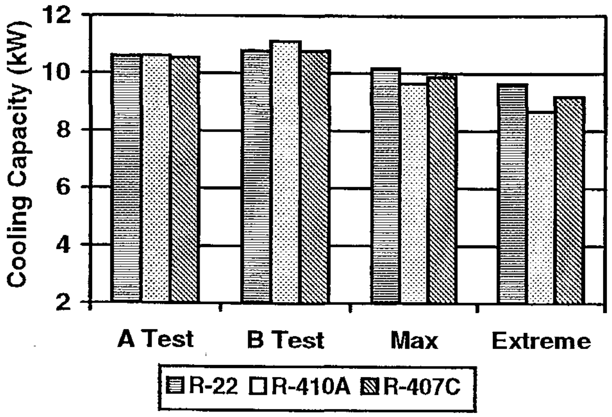

Figure 1. Cooling Capacity

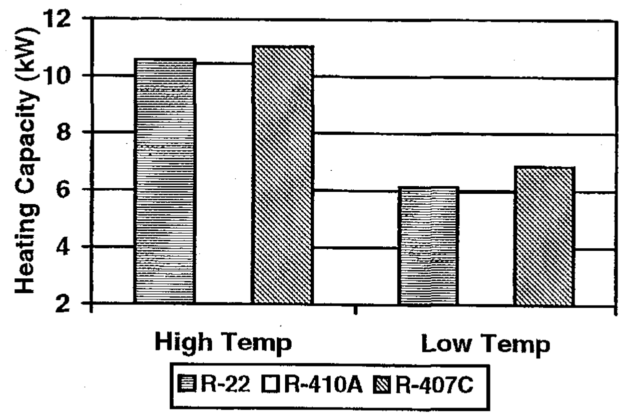

Figure 2. Heating Capacity

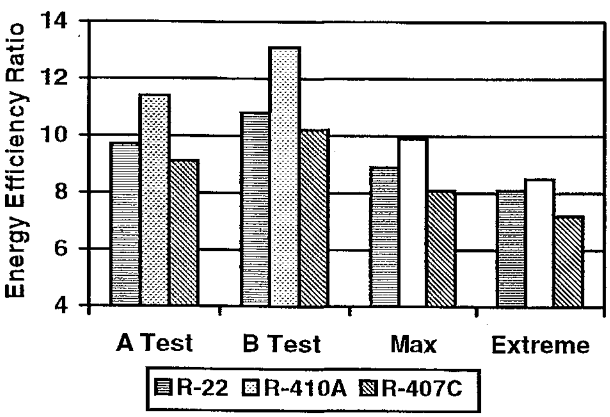

Figure 3. Energy Efficiency Ratio

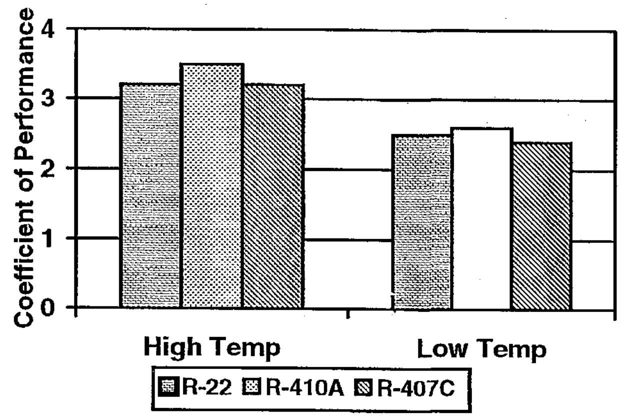

Figure 4. Coefficient of Performance

Cooling Capacity

Figure 1 shows the heat pump cooling capacity for the four air-conditioning test conditions. The experimental values of cooling capacities were obtained from air enthalpy measurements between the inlet and outlet of the indoor unit. Figure 1 shows that at the "A" test condition all the cooling capacities are about the same (compressor speed was changed to provide the same cooling capacity at this condition). R-407C had the same cooling capacity as R-22 at the "B" test condition, the capacity then decreased relative to R-22 to be about 5% lower at the extreme operating condition. R-410A had a 3% higher cooling capacity at the "B" test condition and then the capacity decreased to 5% lower than R-22 at the maximum operating condition and 10% lower at the extreme operating condition.

Heating Capacity

Figure 2 shows the heat pump heating capacity for the high and low temperature test conditions. The experimental values of heating capacities were also obtained from air enthalpy measurements between the inlet and outlet of the indoor unit. The Figure shows that R-407C had a heating capacity that was 5% higher than R-22 at the high temperature test condition and was 12% higher than R-22 at the low temperature test condition. The heating capacity of R-410A ranged from 1% to 2% lower than R-22.

Energy Efficiency Ratio (EER)

The heat pump energy efficiency ratio (EER) is shown for the four cooling test conditions in Figure 3. The EER was derived by dividing the cooling capacity (measured on the airside) by the shaft power input to the compressor plus the indoor and outdoor fan power. R-407C had an EER that ranged from 5.7% to 11% lower than R-22. The EER of R-410A ranged from 4.3% to 21.4% higher than R-22. Compared to the other two refrigerants the EER of R-410A also decreased much more rapidly at the high and extreme operating temperature test conditions. This is due to the lower critical temperature of R-410A. The EER values shown for R-410A do not include any correction factor to compensate for the improved compressor efficiencies at the lower operating speeds.

Coefficient of Performance (COP)

The heat pump coefficient of performance (COP) for the two heating test conditions is shown in Figure 4. The COP was derived by dividing the heating capacity (measured on the airside) by the shaft power input to the compressor plus the indoor and outdoor fan power. R-407C had a COP that ranged from the same as R-22 at the high temperature test condition to 1.6% lower at the low temperature test condition. The COP of R-410A ranged from 7% to 8.4% higher than R-22. The COP values shown for R-410A do not include any correction factor to compensate for the improved compressor efficiencies at the lower operating speeds.

CONCLUSIONS

A comparison was made of the performance of long term replacements R-407C and R-410A with the reference case R-22 in a 10.5 kW (3.0 TR) residential size central heat pump. The performance was measured using the same open drive-reciprocating compressor. The compressor speed was adjusted for each of the two refrigerants to provide the same cooling capacity as the R-22 base case at the "A" test condition. For the R-22 and R-407C performance tests the only other change made to the heat pump was the installation of electronic expansion valves in place of the factory supplied fixed orifice.

R-407C had the same cooling capacity as R-22 at the "B" test condition, the capacity then decreased relative to R-22 to be about 5% lower at the extreme operating condition. R-410A had a 3% higher cooling capacity at the "B" test condition and then the capacity decreased to 5% lower than R-22 at the maximum operating condition and 10% lower at the extreme operating condition.

R-407C had a heating capacity that was 5% higher than R-22 at the high temperature test condition and was 12% higher than R-22 at the low temperature test condition. The heating capacity of R-410A ranged from 1% to 3% lower than R-22.

The heat pump cooling energy efficiency ratio (EER) of R-407C ranged from 5.7% to 11% lower than R-22. The EER of R-410A ranged from 21.4% to 4.3% higher than R-22. Compared to the other two refrigerants the EER of R-410A also decreased much more rapidly at the high and extreme operating temperature test conditions. This is due to the lower critical temperature of R-410A

The heat pump heating coefficient of performance (COP) for R-407C ranged from the same as R-22 at the high temperature test condition to 1.6% lower at the low temperature test condition. The COP of R-410A ranged from 7% to 8.4% higher than R-22. The EER and COP values shown for R-410A do not include any correction factor to compensate for the improved compressor efficiencies at the lower operating speeds.

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